PATENT SPECIFICATION

805,370

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COMPLETE SPECIFICATION

A Slant-Shaft Rotary Piston Engine

We, NSU WERKE AKTIENGESELLSCHAFT, of Neckarsulm/Württemberg, Germany, a German company, and FELIX WANKEL, of Bregenzerstrasse 82, Lindau-Bodensee, Germany, of German nationality, do hereby hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The invention consists in a rotary piston engine having two or more working chambers both or all of which are bounded solely by an inner and an outer spherical surface which are concentric with one another and by conical surfaces on two rotors which revolve continuously and unidirectionally at different speeds having a fixed ratio to one another each about a respective axis, the two axes being inclined to one another and stationary relatively to one another and intersecting at the centre of curvature of the spherical surfaces.

In a preferred form of construction the cones are non-circular, their surfaces being of undulatory conformation so that several generatrices of one cone are in continuous sliding abutment with the surface of the other.

The said form of construction of the rotary piston engine according to the invention provides the basis for a further essential feature of the invention which consists in that one of the conical surfaces is that of a spherical trochoidal cone the characteristics of which depend upon the angle formed by the two cone axes and the relative speeds of revolution of the cones about the said axes, and that the associated cone is represented by or approximates the corresponding conical enveloping surface.

The above described conformation of the two cones enables abutment between them to be along definite fixed generatrices or edges of one of the cones so that narrow radial sealing strips may be provided along these lines. The presence of these strips is essential to enable the engine to function and widens the limits of accuracy to which the remaining surface elements must conform. Normally such radial sealing strips will have a width of a few millimetres. To ensure that these sealing strips actually make contact with the cooperating surface across the whole of their width and thereby reduce wear to a minimum it is preferred to modify the spherical boundary of the trochoidal cone by the generation of a parallel curve of the spherical trochoid. By suitably selecting the distance between the parallel curve and the original curve allowance can be made for the desired deviation of the sealing edge. Moreover, the spherical trochoid may be replaced by circular sectors of arc. The sealing edges of the associated rotor must then move in the direction of the normal of its surface.

Planar representation of the rotor forms is rendered difficult by the fact that the spherical surfaces and the boundary curves of the trochoidal rotor and the associated rotor are not developable and cannot therefore be commensurably projected on to a plane. However, the generation of the rotor surface can be described as follows:—

In Fig. 1 a first shaft is shown at 1, and 2 is the axis of a second shaft, the axes of the two shafts forming an angle α. The point of intersection 3 of the two axes is the centre of a sphere of radius r, which appears in the drawing as a circle 6. Let it be assumed that a tracing pen 5 is rigidly connected with the second shaft at the end of an arm 4. When the sphere 6 revolves about the axis of the shaft 1 and the tracing pen 5 simultaneously revolves about the axis 2 the pen will trace a spherical curve on the surface of the sphere, and this curve will be unicursal whenever the angular speed ratio between the two shafts is $n_1 : n_2$ where n_1 and n_2 are both positive whole numbers and n_2 exceeds n_1 by 1. These curves traced on the surface of the sphere are spherical trochoids with n lobes. According to the speed ratio n_1/n_2 the single tracing pen 5 may be replaced by a plurality of generating points

[*Price 3s. 6d.*]

rigid with the second shaft, all of which will trace out the same curve.

Now let it be assumed that all the points of the spherical trochoid traced out on the surface of the sphere are joined by straight lines with the centre of the sphere. These lines will then determine an undulating conical surface provided on a trochoidal rotor in accordance with the present invention. The form of the surface of an associated rotor between the abutting elements is that of the bevel gear which meshes with the trochoidal rotor (i.e. the envelope of the trochoidal rotor). According to the desired compression ratio the surface of the associated rotor may be made to withdraw from the flank of the bevel wheel to a greater or lesser extent.

Figs. 2 to 6 are examples of various forms and phases of the surfaces of such spherical trochoidal rotors and their associated rotors as used in slant-shaft rotary piston engines of which embodiments are described hereinafter with reference to Figs. 7 and 8, it is to be understood that these figures simply illustrate the form of and manner of co-operation between the conical surfaces, and are not actual illustrations of the rotors themselves. The numerical references conform with the references used in Fig. 1. However, the arm 4 with the tracing pen rigidly connected with axis 2 is replaced by an anti-sphere 4 and contact points at 5.

Fig. 2 represents a pair of rotor surfaces enclosing three working chambers, for operation at a speed ratio of 2:3, shown in such a relative position that the working chamber having the maximum volume apears on the left. In this phase the two other working chambers, only one of which appears in the drawing, enclose intermediate volumes.

Fig. 3 represents the same pair of surfaces in the same phase, the working chamber of greatest volume being seen from the front.

Fig. 4 again shows a phase of the same pair of surfaces where the working chamber with the smallest volume appears on the left, whereas

Fig. 5 represents the phase shown in Fig. 4 viewed from the side. A comparison between maximum working chamber volume (Fig. 3) and minimum working chamber volume (Fig. 5) illustrates the considerable compression ratio that can be achieved.

Fig. 6 represents a similar pair of surfaces designed to a speed ratio of 1:2. The spherical trochoid is located on the surface of a sphere 6 which revolves about the axis 1 whereas the associated rotor surface 4 and the generating point 5 revolve about the axis 2. The illustration shows the greatest chamber volume on the left and the smallest chamber volume on the right.

Fig. 7 shows a practical form of construction of a slant-shafted rotary piston engine according to the present invention, the engine being designed to a speed ratio between the two rotors of 2:3. The shaft of the first rotor is again indicated by 1, that of the second rotor by 2 and the centre of the sphere by 3. 6 represents an outer spherical shell of the first rotor, and 7 a concentric inner spherical surface thereof. The angle between the two shafts is α. In the position shown in the drawing the upper chamber 8 encloses maximum volume. The inner bevel gear annulus 9 creates the differential speeds of revolution of the two rotors. The casing 10 which is indicated diagrammatically contains the two bearings.

The admission and exhaust ports for the working medium are not shown. They may be provided in one of the rotors, on the inner or outer spherical surface, or in the surfaces of the cones.

The channel sections and the port apertures, as well as the timing, depend upon the purpose for which the engine is intended.

The position of the control openings in the spherical surfaces is determined by the purpose for which the engine is used. It depends on whether the engine is designed as a pump for incompressible media, a compressor or a four-stroke engine. For instance a rotary piston engine which is constructed in accordance with the present invention and in which the trochoidal rotor has the higher speed of rotation will be able to produce a four-stroke control solely by the relative movements of the two rotors.

WHAT WE CLAIM IS:—

1. A rotary piston engine having two or more working chambers both or all of which are bounded solely by an inner and an outer spherical surface which are concentric with one another and by conical surfaces on two rotors which revolve continuously and unidirectionally at different speeds having a fixed ratio to one another each about a respective axis, the two axes being inclined to one another and stationary relatively to one another and intersecting at the centre of curvature of the spherical surfaces.

2. A rotary piston engine as claimed in Claim 1, in which at least one of the spherical surfaces is rigid in rotation with one of the rotors.

3. A rotary piston engine as claimed in Claim 2, in which both spherical surfaces are rigid in rotation with one and the same rotor.

4. A rotary piston engine as claimed in any of the preceding claims, characterised in that the fixed ratio of the speeds of revolution of the two rotors is $n_1:n_2$, where n_1 and n_2 are both positive whole numbers and n_2 exceeds n_1 by 1, and that the surface of one of the rotors has the form of a spherical trochoidal cone whereas the other rotor has at least one abutting edge in continuous sealing contact with said surface.

5. A rotary piston engine as claimed in Claim 4 modified in that the surface of the

one rotor has the form of a parallel surface to a spherical trochoidal cone and the abutting edge or edges of the associated cone are each provided with a radially disposed sealing strip in continuous sealing contact with said surface.

6. A rotary piston engine as claimed in Claim 4 or 5 characterised in that the surface of the trochoidal rotor consists by way of approximation of sections of circular cones and that the abutting sealing strips on the associated rotor extend along generatrices of the conical surfaces and are slightly movable in a direction at right angles to said generatrices.

7. A rotary piston engine as claimed in any one of Claims 1 to 6 characterised in that thin walled and elastically yielding sealing elements are interposed between the abutting conical elements and/or the truncated edges of the cones and the inner and outer spherical surfaces.

8. A rotary piston engine as claimed in any one of the preceding claims characterised in that gearing between the two rotors consists of an internal bevel gear annulus and a cooperating bevel gear.

9. A rotary piston engine as claimed in any one of the preceding claims characterised in that the opening and closing of the channels and ports for the working medium is effected by the relative movement of the inner and/or the outer spherical surfaces.

10. A rotary piston engine as claimed in any one of the preceding claims characterised in that the cone surfaces of both rotors are of generally undulatory conformation so that several fixed generatrices of one rotor are in permanent sliding abutment with the surface of the other rotor.

11. A rotary piston engine substantially as hereinbefore described and illustrated in Fig. 7 of the accompanying drawings.

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805,370 COMPLETE SPECIFICATION
3 SHEETS

This drawing is a reproduction of the Original on a reduced scale.
SHEET 1

Fig. 1

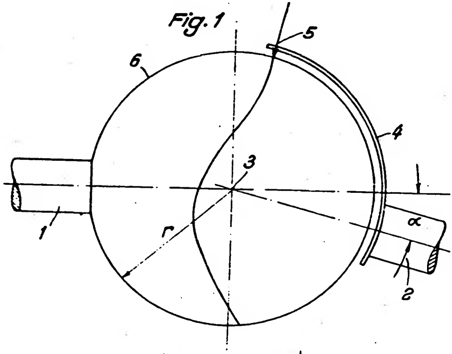


Fig. 2

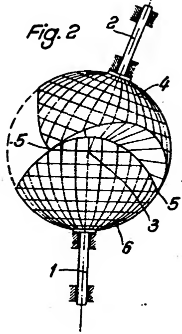


Fig. 3

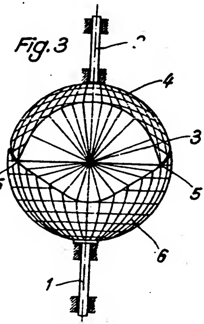


Fig. 4

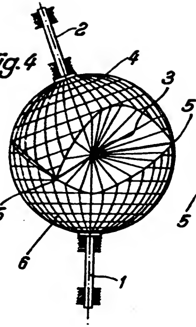


Fig. 5

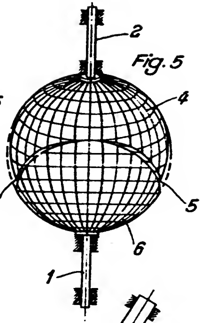
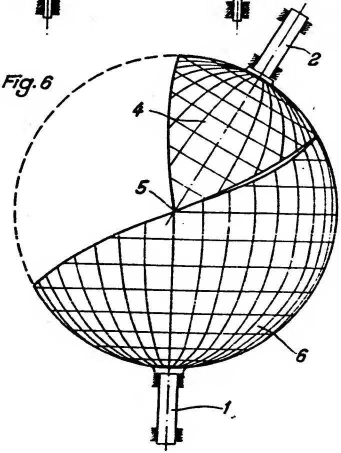
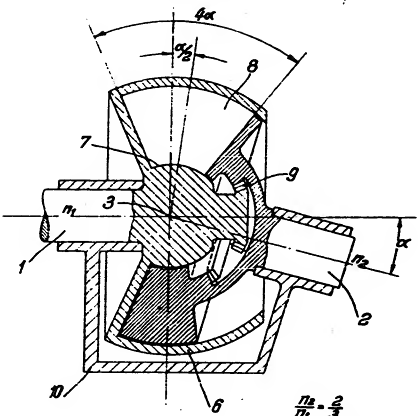


Fig. 6



805,370 COMPLETE SPECIFICATION
3 SHEETS This drawing is a reproduction of the Original on a reduced scale.
SHEETS 2 & 3

Fig. 7



$\frac{n_2}{n_1} = \frac{2}{3}$
$\alpha = 15°$

1,099,085

PATENT SPECIFICATION

DRAWINGS ATTACHED

1,099,085

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Int. Cl.:—F 01 c 3/00.

COMPLETE SPECIFICATION

Rotary Positive Displacement Units

I, ERNEST WILDHABER, a citizen of the United States of America, of 124 Summit Drive, Rochester, New York, United States of America, do hereby declare the invention for which I pray that a patent may be granted to me, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to rotary positive-displacement units. The unit may be embodied as a compressor, a pump, a motor or engine, and as a unit combining a compressor and a motor. A unit referred to comprises at least a housing, a pair of rotors with interengaging lobes or teeth designed for mesh all around their periphery, and ducts for leading fluid to and from the rotors. The pair of interengaging rotors have numbers of lobes or teeth that differ by one.

Pumps of this general description are known with rotors that run on parallel axes, and where one rotor has teeth provided on its outside periphery while the other rotor surrounds said one rotor and has teeth provided internally thereon. Such known structure is limited in tooth depth, which is bound to be the circular pitch divided by $\pi = 3.1416$. This limits the displacement, and there are other limitations.

One object of the present invention is to devise a unit of the character referred to which is not thus limited, and where the lobe depth is larger than one third of the circular pitch, usually much larger. A further object is to devise a unit whose rotors can run at very high speed and which can stand far more centrifugal inertia loads than the internally toothed member of known design. A further aim is to devise a unit with a pair of rotors which expand more nearly in a direction lengthwise of the lobes when run at high temperature, so as to remain well matched.

In the known rotors with parallel axes, contact is mainly with the sharply curved end surfaces of the teeth. These contact side surfaces or each other. A further and important aim is to devise a unit wherein the side surfaces themselves of the mating lobes are adapted to contact each other, and to contact so as to have a duration of contact between side surfaces larger than one pitch. These mating side surfaces are only moderately curved away from one another and can carry considerable load. A still further aim is to devise a unit for gases or air, that can be run at very high speed without requiring timing gears, and where the lubrication provided on the side surfaces of the lobes by the gaseous fluid is sufficient to carry the loads sustained.

A still further object is to provide a unit embracing a compressor and a motor or engine of different volume displacement, where one of the pair of rotors of the compressor is rigid with one of the pair of rotors of the motor.

Other objects will appear in the course of the specification and in the recital of the appended claims. These objects may be attained singly or in any combination. They are attained with rotors whose axes intersect at an angle.

In the accompanying drawings:

Fig. 1 is a diagram explanatory of the principles underlying the present invention.

Fig. 2 is an axial section of a unit constructed according to the present invention.

Fig. 3 is a section through the casing of this unit, taken along lines 3-3 of Fig. 2.

Fig. 4 is an axial section of a modified unit.

Fig. 5 is an axial section, partly a side

view of a rotor of modified design.

Fig. 6 is an axial view of a rotor constructed according to the present invention, looking at its face, showing its straight lobes or teeth directed towards its apex.

Fig. 7 is a similar axial view, showing lobes curved lengthwise.

Fig. 8 is an axial view of a modified rotor showing lobes curved lengthwise and spirally inclined.

Fig. 9 is a diagram illustrating one way of producing rotor lobes or teeth constructed according to the invention. It shows the rotor in axial section.

Fig. 10 is a similar diagram employing reciprocating tool motion.

Fig. 11 is an axial section of a rotor and a diagrammatic view of tool means corresponding to Fig. 10, looking in the direction of the tool motion. The tool is shown in a position cutting or grinding the lobe bottom.

Fig. 12 is a view looking in the direction of the tool motion when the tool engages the side of a lobe. Fig. 12 corresponds to Figs. 10 and 11 and shows the rotor in a diagrammatic view in a tilted position.

Fig. 13 is a diagrammatic view like Fig. 12, but illustrating a different tool profile.

Fig. 14 is an enlarged view of the rotors shown in Fig. 2, looking at right angles at the plane of their axes.

Fig. 15 is a fragmentary view thereof taken in the direction of the instantaneous axis (32 in Fig. 2) and showing the fully engaged lobes.

Fig. 16 is a fragmentary view looking along the line of contact of the lobe ends, taken along line 39 of Fig. 2.

Fig. 17 is a diagrammatic view of a unit comprising a compressor and a motor or engine.

Fig. 18 is a diagrammatic section taken along lines 18-18 of Fig. 17.

Figs. 19 and 20 are diagrams illustrating the advantage obtained in inclining the plane of the axes of each of the two rotor pairs with respect to each other. They can be considered sections along lines 19-19 of Fig. 17, looking in the direction of the arrows.

In the diagram Fig. 1, numerals 30, 31 denote the rotor axes intersecting at apex O. The rotor with axis 31 has any number N of lobes or teeth, while the rotor with axis 30 has (N+1) lobes or teeth. As the rotors run at the ratio of their tooth numbers, there is a fixed axis 32 of instantaneous relative motion. It can be considered the line of contact of conical pitch surfaces which move with the rotors and roll on each other without sliding. Any point 33 of the instantaneous axis 32 has distances from the axes 31, 30 in proportion to the tooth numbers N, (N+1).

Let s denote the angle between the axes 30, 31 and p the inclination of instantaneous axis 32 to the vertical. The pitch angles 28, 29 of the two rotors are then $(90°-s-p)$ and $(90°+p)$. The distances of point 33 from the axes 31, 30 can then be expressed as $(0-33)\cdot\cos(s+p)$ and $(0-33)\cdot\cos p$ respectively.

Their proportion $\dfrac{\cos(s+p)}{\cos p}$ equals $\dfrac{N}{(N+1)}=m$.

As known this can be transformed into $\tan p=\dfrac{\cos s-m}{\sin s}$, where $m=\dfrac{N}{(N+1)}$

It will now be shown numerically how much more lobe depth is attainable with intersecting axes than with parallel axes.

For instance at a ratio of the lobe numbers of $(N+1)/N=9/8$, angle p equals 16° 35' at $s=15°$. The tooth depth $2s$ is then 30°, measured on a great circle. The angular pitch, also measured on a great circle is $\dfrac{360°}{9}\cdot\cos p=38°\ 20'$. The proportion of lobe depth to pitch is 30/38.33= 1//1.28.

With parallel axes of the rotors the pitch circle circumferences differ by the circular pitch, and the pitch circle diameters differ by the circular pitch divided by π. The tooth depth equals the difference in pitch circle diameters. The proportion of tooth depth to circular pitch is thus unavoidably $1/\pi=1/3.1416$. It is less than one third.

The depth proportion has a profound influence on the lobe shape attainable, and of course the larger proportion increases the displacement.

The rotors 34, 35 (Fig. 2) with axes 30, 31 are rotatably mounted in a housing 36 formed by two parts tightly bolted together. The rotors have convex outer surfaces lying in a common spherical surface 37 that is centred at apex O at which the axes 30, 31 intersect. This surface matches a counterpart concave spherical surface provided in housing 36. The counterpart surface serves as a seal and is interupted by ducts 38, 38'. When Figs. 2 and 3 refer to a compressor, the outlet pressure is higher than the inlet pressure and the outlet area is more confined. 38 is the inlet, 38' the outlet for a compressor. On a motor or engine the inlet pressure is higher, and the inlet area is more confined. Looking at Fig. 3, rotation is in clockwise direction for a compressor, in counter-clockwise direction for a motor.

At their inner ends the lobes 45, 46 bear against or match a spherical surface por-

tion 40 also centered at O. This portion is formed integral with a cylindrical portion 41 and shaft 42. Rotor 34 is made rigid with shaft 42 by lateral projections that engage slots 43 provided on a flange 44 of shaft 42.

Figs. 14 to 16 afford enlarged views of the rotors 34, 35. Their interengaging lobes 45, 46 form pockets 47, 47′, 47″ ... of changing volume. When rotation is in direction of arrow 48 the pockets diminish in volume. On a compressor the outlet area starts in a region where the volume has been substantially decreased.

Profiles

I may start a design by assuming the lobe numbers N, (N+1) and the angle s between the shafts. This determines the angular lobe depth, which is $2s$. Next I may determine the axes of curvature of the convex outer lobe ends, and particularly their angular distance from each other when they contact at their outer ends (Fig. 16). The lobe surfaces now considered are composed of st aight-line elements that all pass through apex O. The end surface portion lies either in a circular conical surface or in a surface whose curvature at the end coincides with that of a conical surface. In one embodiment, the end-surface portion $46e$ (Fig. 14) of rotor 35 having N lobes is part of a circular conical surface. Fig. 1 shows the axis 50 of this conical surface in the position where it lies in the plane of the axes 30, 31. It includes an angle b with the instantaneous axis 32. 50′ shows the position of axis 50 after rotation about rotor axis 31 through half a turn, so that it again lies in the plane of the axes 30, 31, the drawing plane of Fig. 1. The curvature axis 51′ of the lobe end $45e$ of rotor 34 includes an angle or arc b' with curvature axis 50′. This arc is preferably smaller than half the circular pitch, measured as an arc on a great circle whose plane passes through O.

The following formula can be derived for the angle b' with the methods of spherical trigonometry:

$$\tan b' = \frac{P \sin^2 c}{1 + P \sin c \cos c}$$

wherein $P = \tan (s+p) - \tan p$

and $c = 2(s+p) - b$

Angle b is changed until b' attains a desirable amount.

After assuming the lobe end $46e$ of rotor 35 and its axis of curvature 50 the entire surface of the lobes 45 can be determined as the surface enveloped by this lobe end when it is turned about axis 31 while rotor 34 turns on its axis at the ratio of the tooth numbers or lobe numbers (N+1) and N, so that the conical pitch surfaces roll on each other without sliding.

When the assumed tooth end $46e$ is a circular conical surface, it can be embodied by a conical milling cutter or grinding wheel whose axis coincides with axis 50. When the axis of this tool is moved relatively to a rotor 34 as stated above, the rotating tool will produce the lobes thereof.

During such motion axis 50 (Fig. 1) describes a conical surface, and its point 53 traces a circle 53′ showing up as a straight line. Contact between rotor 34 and the conical end surface of rotor 35 is always in a plane that connects the instantaneous axis 32 with the considered position of axis 50. It is along the cone element that lies in said plane, the normal plane of said element. Broadly contact is in a turning position where the normal plane of a considered straight-line element passes through and contains the instantaneous axis. The said plane is then a contact plane, common to both rotors.

The desired tooth thickness is attained by changing an initial assumed cone angle of the end surface.

It should be noted that the side surfaces 45_s have a desirable moderate inclination to axial planes, that is to planes containing axis 30. This is a result of the increased lobe depth. The side profiles, the profiles of the side surfaces 45, are inclined less than thirty degrees to the axial plane (49, Fig. 15) passing through their mid-portion.

The side surfaces 45, contact the end surfaces $46e$ of rotor 35, and they have a further position of contact. The said contact plane passes through the instantaneous axis 32 a second time in its rotation about axis 30. In the second position contact is between the lobe sides 45, 46, of the rotors 34, 35. This is also apparent from consideration of Fig. 15. Obviously the sides 45, have mating lobe sides 46,. The contact position for the sides can be computed with the known procedures of spherical trigonometry. Knowing the contact position and the line of contact the mating shape itself can be readily determined. Line 54 (Fig. 15) is the path of contact between the lobe sides 45, 46,. Line 54′ is the path of contact for the opposite lobe sides. It is seen that the duration of contact obtained with path 54 is larger than one full pitch, as the lobes are relatively deep. Always at least one pair of lobes contact with their side profiles, on each of the two opposite lobe sides. Because of the moderate relative curvature of the side profiles a considerable load can be carried at all times, even under adverse conditions of lubrication, or with lubrication by gaseous fluid only.

Dotted line 54, is the path of contact of the lobe ends 46,.

33-33,, Fig. 2, is the lobe addendum of rotor 34; and 33-33, is the lobe dedendum

thereof. While the addendum is smaller than the dedendum in the example illustrated, it may also be made larger than the dedendum in many cases.

In the embodiment of Figs. 2 and 3 the outlet opening and the inlet opening are both provided adjacent the outer end of the rotor lobes, at the spherical surface 37.

Modified Embodiments

Fig. 4 illustrates an embodiment with inlet opening at the inner spherical surface 40 and outlet opening at the outer spherical surface 37. This embodiment may be used, for instance, on pumps, where the centrifugal action adds to the effect. The rotors 34', 35' with axes 30, 31 have the described shape. They are rotatably mounted in a housing 55 to which is rigidly secured a tubular member 56 with rounded end 56'. This end contains the described inner spherical surface portion 40 on its outside and an adjacent cylindrical surface portion 41' which serves as a journal for rotor 34'. Fluid enters along axis 30 and through openings 57 provided in end 56'. Outlet duct 58 leads to an outlet not shown.

In the embodiments so far illustrated the inner spherical surface 40 has about half the diameter of the outer spherical surface 37. If the lobes or teeth were continued all the way to apex O, the volume displaced inside of surface 40 would be only one eighth of the volume displaced inside of surface 37 of double diameter. With lobes having their inner ends at surface 40 the displacement is still 7/8 of the maximum displacement possible inside of surface 37, while a material advantage is gained in the production by cutting or grinding. However when production is for instance by moulding, coining or casting the production advantage is reduced and I may diminish the inner spherical end surface to almost nothing. Fig. 5 shows a rotor 34" so shaped. The inner spherical surface 40, has less than half the radius of the outer spherical surface 37, and the knob containing surface 40, is formed integral with rotor 34".

Proportionately small inner spherical surfaces may also be provided on rotors of large size, and on rotors intended for extreme rotational speeds where the increased production cost is acceptable.

Fig. 6 is an axial end view of a rotor 34, that may be used as rotor 34 in the described units. It has straight lobes or teeth 60 directed towards apex O, at which the axes of the rotor pair intersect. The surface of its lobes is made up of straight-line elements, all passing through apex O.

Fig. 7 is an axial view of a rotor 34, whose lobes 60, are curved lengthwise, chiefly for a production advantage in medium and small sizes. Opposite lobe sides 61, 61' are convex and concave lengthwise, respectively. At their mid-portion the lobes are directed towards apex O.

The rotor 34, shown in Fig. 8 has longitudinally curved lobes 60, that are spirally arranged. Spirally arranged lobes may be considered especially in the embodiment described with Fig. 4. The types illustrated in Figs. 6 to 8 can each be used in both described embodiments.

If desired, I may use blades or vanes in the inlet and outlet passages, to direct the fluid more positively.

Production

One way of producing a rotor 34 with straight radial lobes in a generating operation has already been described. There a conical milling cutter or grinding wheel describing (reproducing) a tooth end of rotor 35 is caused to roll on rotor 34 as if it were a part of rotor 35. The mating rotor 35 may be cut with the same conical milling cutter as used in cutting rotor 34 in a similar rolling motion. Here however the turning motion about the two intersecting axes is at a varying ratio. A timing change is introduced for instance by cam means.

Another method, also suited for large sizes, will now be described with reference to Fig. 9. It uses a conical milling cutter 62 or grinding wheel similar to that used in cutting rotor 34. The axis 63 of the cutter used intersects the axis 30 of the rotor blank 34, at apex O. The cutter 62 is rotated on its axis for stock removal while the workpiece 34, is slowly turned on its axis 30 and while the cutter is fed about apex O. It is fed about an axis intersecting the workpiece axis at apex O and that may be perpendicular to axis 30 and to the drawing plane. Another position of cutter 62 during its feed motion is shown in dotted lines 62'. The feed motion of the cutter is at a varying ratio to the turning motion of the workpiece. It is a slow oscillating motion between two end positions timed to the turning motion of the workpiece. While the turning motion of the workpiece can be at a uniform rate, a better production rate is attained by cyclically varying the speed of the feed motion.

The mating rotor 35 may be produced with the same cutter and the same method, using a different timing between the turning motion of the workpiece and the angular feed motion between the cutter and workpiece.

Figs. 10 to 12 illustrate a method using tool reciprocation. Edged tool 64 or grinding wheel 64' has a convexly curved working profile 65 or cutting edge, that is at least as much curved as the concave bottom profile of the rotor at the inner lobe end. It is a circular arc centered at 67 (Fig. 11). The sides 68 merely serve to support the

working profile without contributing to the shape produced. Tool 64 or wheel 64' is reciprocated lengthwise of the lobes towards and away from apex O along line 70. Line 70 lies in the plane of Fig. 10 and is perpendicular to the drawing plane of Figs. 11 and 12. Figs. 10 and 11 show the position where the tool cuts the lobe bottom along line 70. To produce the side surfaces of the lobes, the workpiece 34 is fed angularly about an axis 59 passing through apex O and perpendicular to the drawing plane of Fig. 10. Dotted lines 34' show the workpiece in a tilted position where a straight-line element of the lobe side is produced. This cutting position is also shown in Fig. 12.

To cut or grind along a fixed line 70 all the time, the cutting tool or grinding wheel is fed in a circular path 72 about an axis coinciding with line 70, at such a rate that the profile inclination of the tool at line 70 coincides continuously with the profile inclination of the workpiece. This additional feed motion of the tool may be a circular translation, where the tool remains parallel to itself, and where center 67 describes arc 73. The said additional motion is timed with the turning motion of the workpiece. The angular feed motion between the workpiece and tool is also timed with said turning motion, again at a varying rate.

Production advantages are gained by periodically varying the speed of the feed motions. The process is continuous. The workpiece is fed in and out about axis 59 once for each lobe. The lobes are completed after the workpiece has made a full turn.

Grinding wheel 64' should be small enough to clear the workpiece 34 on the far side. The size restriction is less severe when the larger rotor 34 of the pair has an uneven number of lobes. Then a lobe space is diametrically opposite a lobe; and the grinding wheel can reach somewhat into a lobe space when grinding the outside end of a lobe.

Roughing may be accomplished with a rotary cutter 64" (Fig. 10) of the same profile shape, without reciprocation but with the same feed motions.

Spiral teeth may be produced by adding a continuous turning motion of the workpiece and timing the reciprocations with said added motion.

Fig. 13 illustrates a working profile 73 of varying curvature. It is convex and most curved at its end 74 and less curved on both sides 75. It can be used on a reciprocating tool or grinding wheel or also on a tool 76 that oscillates about an axis 77 for cutting. In the latter case the mating rotor 35 with convex face surface may be cut with a rotating cutter that turns about axis 77.

Axis 77 lies in a plane perpendicular to line 70. The said additional tool motion is in this plane and is such that the tool profile 73 always passes through the fixed point at which line 70 intersects said plane. This additional tool motion is so timed with the turning motion of the workpiece that the tool profile has the same inclination as the lobe profile passing through said fixed point, so that the two profiles contact at said point. Except for the additional tool motion, the feed motions are the same as in the method described with Figs. 10 to 12. They are such that fixed line 70 describes the lobe surfaces.

While the additional tool motion for working profiles (73) of varying curvature is more complex than for circular arcuate working profiles, there is less tool wear, improved finish of the lobe sides and improved production efficiency.

The described lobe shape is furthermore singularly well-adapted to finishing by shaving, burnishing and lapping because of the many simultaneous contacts all around the circumference.

Multiple Rotor Pairs

Diagram Fig. 17 shows a disposition of a unit with two pairs of rotors 80, 81 and 82, 83 of different volumetric displacement rotatably mounted in a housing 84. It can be embodied as an engine unit or also as a cooling unit. In an engine unit the rotors 80, 81 are part of a compressor. Duct 85 admits air to the compressor. Duct 86 leads the compressed air to a combustion chamber diagrammatically indicated at 88. Duct 89 leads the combustion gas to the other pair of rotors 82, 83. An exhaust duct 90 leads the expanded combustion gas therefrom to the outside.

The angularity of the rotational axes 91, 92 of rotors 80, 81 is smaller than the angularity of the axes 92, 93 of the rotors 82, 83. The rotors 81, 82 have a common axis 92 and are rigid with each other. Furthermore the plane of the intersecting axes 91, 92 is preferably inclined to the plane of the intersecting axes 92, 93 to achieve improved pressure balance between the rotors 81, 82. The reason for this inclination will now be shown with Figs. 19 and 20.

Fig. 19 illustrates the load conditions when the axes 91, 92, 93 are all in one plane. The compressed air exerts a one-sided pressure on rotor 81. This pressure may be along an arrow 94 shown in Both Fig. 17 and Fig. 19. It intersects the mid-plane 78 of shaft portion 78' at a point 79. A similar one-sided pressure is exerted on rotor 82. It may extend in the direction of arrow 95 that intersects plane 78 at 96. The length of the arrows are measures of the loads. The axial pressure components through points 79, 96 partly balance each other, but they also produce a turning moment in a

plane parallel to axis 92 containing points 79, 96. The turning moment increases with increasing distance of the points 79, 96 from one another. The load components in mid-plane 78 can be combined to a resultant 92-97 (Fig. 19) which passes through axis 92 plus a turning moment about said axis. The turning moment is the difference of the turning moments exerted by the loads passing through the points 79, 96.

Fig. 20 shows the effect of a displacement of the plane of the axes 91, 92 about axis 92. Displacement is through an angle which moves point 79 to a position 79' close to point 96. The resultant load 92-97' in mid-plane 78 is slightly smaller than load 92-97. But the main gain lies in the sharp decrease of the turning moment exerted by the axial load components passing now through points 96, 79', because of the proximity of these points. Diagram Fig. 17 should be understood to correspond to the showing of Fig. 20.

The resulting turning moment about axis 92 is relatively small. The torque exerted on rotor 82 of the motor drives the compressor. Only a moderate portion thereof is transmitted through the lobe sides in the rotor pair 82, 83.

In operation the air is compressed in the rotor pair 80, 81 to a degree controlled by edge 87 (Fig. 17) of the outlet opening. This pressure is maintained in the combustion chamber, where combustion expands the volume of the gaseous fluid. The rotor pair 82, 83 thus requires a larger displacement volume, which is built into the diameter of the rotors and into the angularity of their axes.

When the amount of fuel injected in to the combustion chamber is diminished under the design amount, the expansion through combustion decreases, and the pressure in the combustion chamber decreases. The compressed air then expands after leaving the compressor, increasing its volume, until the further increase in volume through combustion equals the volume displaced in the rotor pair 82, 83.

A gain in efficiency at reduced or changed engine load can be achieved by changing the outlet opening of the compressor for different loads, increasing the outlet area when fuel admission is decreased. This changes the pressure directly, without requiring expansion of the compressed air when leaving the compressor.

Figs. 17 and 18 diagrammatically illustrate means for changing the outlet opening of the compressor. Edge 87 of the outlet opening is provided on a part 98 that can be angularly adjusted about axis 91 in dependence of the fuel admission. Dotted lines 98' show part 98 after adjustment for a decrease in compression and decrease in fuel admission. Edge 87 is then at 87'. The displacement of the opposite end 98, has little effect on the performance, because there is practically no change in volume in that region.

Cooling of the hot parts may be provided in any suitable known way.

When the unit shown in Fig. 17 is embodied as a cooling unit, the larger rotor pair 82, 83 takes the incoming fresh air and compresses it. The air thereby heats up and is transmitted to a cooler, or broadly to a thermal part 88 for changing the temperature of the gaseous fluid. After cooling it is led to the rotors 80, 81 which now serve as a motor. As it expands therein the temperature drops to the level for which the unit was designed. A stream of cool fresh air leaves the unit.

WHAT I CLAIM IS:—

1. A positive displacement unit usable, for example, as a compressor, motor or pump, comprising a housing, a pair of rotors rotatably journaled in said housing and having interengaging lobes or teeth, which mesh around the whole of the peripheries of the rotors, the number of lobes or teeth in the two rotors differing by one, and the rotary axes of the two rotors being inclined to one another.

2. A positive displacement unit as claimed in claim 1, wherein the depth of the lobes or teeth of each rotor in an axial plane of each rotor is larger than one third their circular pitch.

3. A positive displacement unit as claimed in either claim 1 or claim 2, wherein the teeth of the rotors are straight and converge to the point of intersection of the rotor axes.

4. A positive displacement unit as claimed in any of the preceding claims, wherein the rotors have convex spherical outside surfaces which lie in a common convex spherical surface, the housing has inlet and outlet openings, and at least the outlet opening lies on this common spherical surface.

5. A positive displacement unit as claimed in claim 4, wherein the inlet opening also lies on said common spherical surface.

6. A positive displacement unit as claimed in claim 4, wherein the rotors have also spherical concave surfaces which lie in a common inner spherical surface, and the inlet extends through this inner spherical surface.

7. A positive displacement unit as claimed in any of the preceding claims, wherein the lobes or teeth of both rotors have rounded tops adapted to come into contact, and the radius of curvature at the center of the tooth tops of both rotors is less than half the circular pitch.

8. A positive displacement unit as

claimed in claim 7, wherein the lobes or teeth of one rotor, which is smaller, have a conical outer surface whose axis passes through the point of intersection of the rotor axes.

9. A positive displacement unit as claimed in any of claims 4 to 8 inclusive, wherein the inlet opening is adjustable.

10. A positive displacement unit as claimed in any of the preceding claims, wherein the lobes or teeth of the rotor pair are curved lengthwise.

11. A positive displacement unit as claimed in any of the preceding claims, wherein the lobes or teeth have side portions inclined to axial planes of the respective rotors by less than thirty degrees and more curved top portions and bottom portions, said lobes or teeth being deep enough so that their mating side portions have a duration of contact in excess of one pitch.

12. A positive displacement unit as claimed in any of the preceding claims wherein one rotor has a pitch angle larger than a right angle, and the other rotor has profiles with convex circular arcuate tops

13. A positive displacement unit as claimed in any of the preceding claims, wherein there are two pairs of rotors such as claimed, and one rotor of one pair is coaxial with one rotor of the other pair and is fixedly secured thereto.

14. A positive displacement unit as claimed in claim 13, wherein the intersecting axes of one pair of rotors is inclined to the plane of the axes of rotation of the other pair.

15. A positive displacement unit as claimed in claim 13, wherein the intersecting axes of one pair of rotors includes an angle which is different from the angle included between the axes of the other pair.

16. A positive displacement unit as claimed in any of claims 13 to 15 inclusive, wherein the two pairs of rotors have different volumetric displacement, the rotor pair of smaller displacement is part of a compressor, a first duct is provided for admitting air to said compressor, a second duct is provided for conducting compressed air from said compressor to a combustion chamber through an opening adjacent the periphery of the compressor rotors, means is provided for changing the opening to control the compressor ratio, a third duct is provided for conducting combustion gas from said chamber to the other pair of rotors, and a fourth, exhaust duct is provided for conducting the expanded combustion gas from said other pair of rotors to the outside.

17. A process for producing a toothed rotor, as claimed in any one of the preceding claims, for a positive displacement unit, comprising rotating a rotor blank continuously on its axis while imparting a working motion to a tool, effecting a relative feed motion between tool and blank about an axis which intersects the axis of the blank, so that the tool moves from the top to the bottom of a tooth and then from the bottom to the top of the tooth.

18. A process as claimed in claim 17, wherein the axis of feed is at right angles to the axis of the blank, and the feed motion takes place from the top to the bottom and then back to the top of each tooth.

19. A process according to either claim 17 or claim 18, wherein the rotation of the blank about its axis is at a varying speed during production of each tooth.

20. A process according to any of claims 17 to 19 inclusive wherein the tool is a milling cutter whose cutting edges lie in a conical surface whose axis coincides with the axis of the cutter.

21. A process according to any of claims 17 to 20 inclusive wherein the tool is a reciprocating tool with a rounded working profile, the feed motion between the tool and blank being varied for each tooth so that a predetermined point of the tool profile describes a mean tooth profile, and displacing the tool during production of each tooth side to keep engaging the tooth profile continuously at said point.

22. A process according to claim 21, wherein the working profile of the tool has a varying curvature, with the largest curvature at the point which comes into contact with tooth bottom.

23. A positive displacement unit substantially as described herein with reference to the accompanying drawings.

24. The process of producing a rotor as claimed in any one of claims 1 to 16, for a positive displacement unit substantially as described herein.

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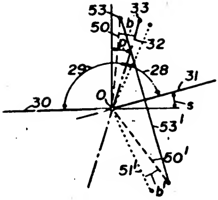

FIG. 1

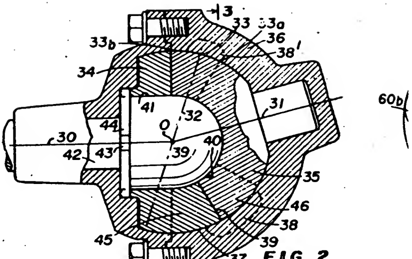

FIG. 2

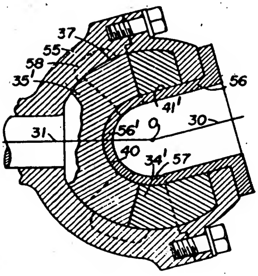

FIG. 4

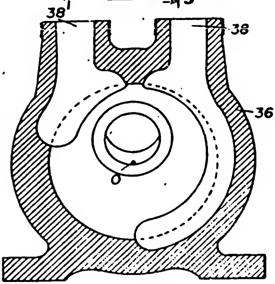

FIG. 3

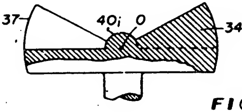

FIG. 5

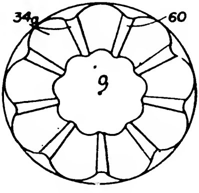

FIG. 6

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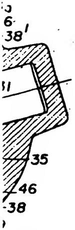
SHEETS 1 & 2



IG. 2

FIG. 3

34''

IG. 5

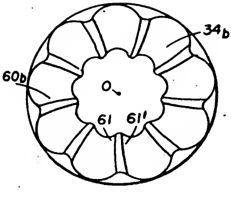


FIG. 7

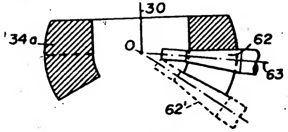


FIG. 9

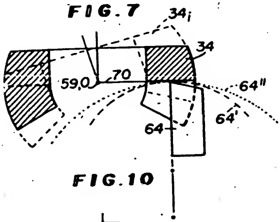


FIG. 10

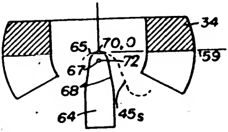


FIG. 11

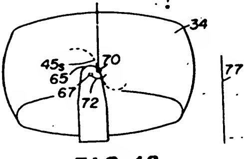


FIG. 12

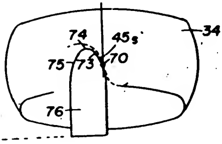


FIG. 13

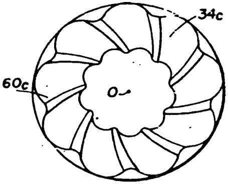


FIG. 8

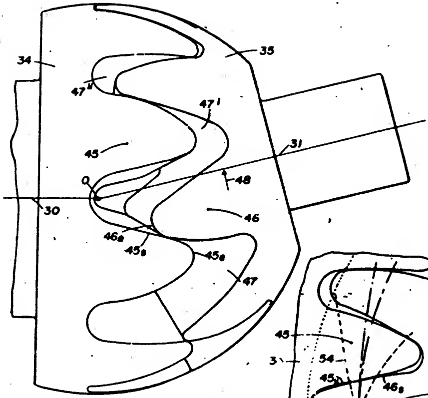


FIG. 14

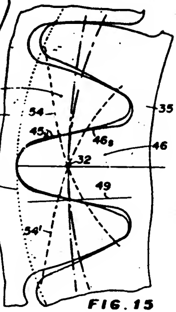


FIG. 15

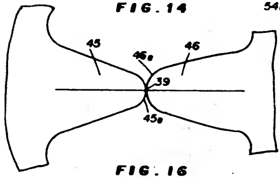
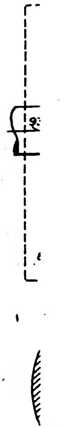


FIG. 16



1,099,085 COMPLETE SPECIFICATION
4 SHEETS This drawing is a reproduction of the Original on a reduced scale.
SHEETS 3 & 4

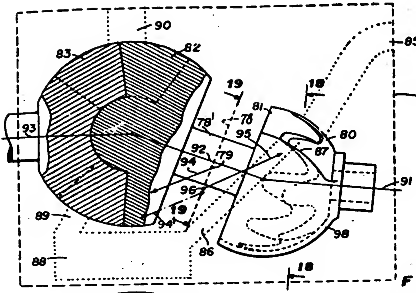


FIG.17

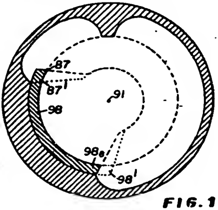


FIG.18

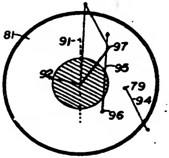
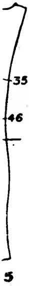


FIG.19



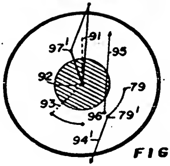


FIG.20